# CFD ANALYSIS OF A VERTICAL TUBE WITH EXTERNAL HELICAL FINS IN NATURAL CONVECTION

A PROJECT REPORT SUBMITTED IN THE PARTIAL FULFILLMENTS OF THE REQUIREMENTS FOR THE DEGREE OF

**B.Tech**

(**MECHANICAL ENGINEERING**)

by

ANIKET NANGIA

Roll number- 108ME060

Under the supervision of

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**May 2012**

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# National Institute of Technology
# Rourkela

## C E R T I F I C A T E

This is to certify that the work in this thesis entitled "**CFD analysis of a vertical tube with external helical fins in natural convection"** by **Aniket nangia,** has been carried out under my supervision in partial fulfillment of the requirements for the degree of **Bachelor of Technology** in ***Mechanical Engineering*** during session 2011- 2012 in the Department of Mechanical Engineering, National Institute of Technology, Rourkela.

To the best of my knowledge, this work has not been submitted to any other University/Institute for the award of any degree or diploma.

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# CONTENTS

# ABSTRACT

Convective heat transfer between a surface and the surrounding fluid in a heat exchanger has been a major issue and a topic of study for a long time. In this study, an attempt has been made to study the effect of various fin configurations on the total heat transfer from a vertical tube. The temperature contours, velocity vectors, surface nusselt number, total heat transfer rate from the fins as well as wall of the tube were calculated and plotted using ANSYS 13.0. The Navier-stokes equations were used to solve for the fluid flow inside the tube and the Boussinesq approximation was used to model the buoyancy effect. Aluminum was used as the base metal for the pipe as well as the fin material. The fluid surrounding the tube was air.

Three different configurations have been used in this analysis. The three configurations were compared on the basis of total heat transfer. It was found that the fin configuration with trapezoidal fins had the most heat transfer rate.

# CHAPTER 1

# 1. INTRODUCTION

Natural convection [1] is a process or type of heat transport, in which the fluid motion is not caused by any external source but only by density differences in the fluid occurring due to temperature gradients. Here the fluid which surrounds a heat source receives heat, becomes less dense and rises. The fluid that is surrounding the hot fluid is cooler and then moves to replace it. This cooler fluid gets heated and the process continues, forming convection current. The driving force for this process is buoyancy, a result of difference in the fluid density. Natural convection has attracted a great deal of attention from researchers because of its presence both in nature and engineering applications.

Fins or extended surfaces are frequently employed to increase the vapour side heat transfer rate in various heat transfer applications [2]. These are primarily used to increase the surface area when the heat transfer coefficient on the fluid side is relatively low so as to enhance the total rate of heat transfer [2]. Since most of the real fins are thin, they are treated as one-dimensional .Fin efficiency is a function of the geometry, material thermal conductivity, heat transfer coefficient of the fin at the fin surface and the fin tip boundary condition [2].

Convection heat transfer [1] is between a surface (at $T_w$) an the surrounding fluid is given by

$$Q=hA(T_w-T_{amb})$$

Where h is the heat transfer coefficient and A is the surface area of heat transfer. The heat transfer rate may be increased by increasing the surface area across which convection occurs. This may be done by using fins that extend from the wall in to the surrounding fluid. A straight fin is any extended surface that is attached to a plane wall. It may be of uniform cross sectional area or its cross sectional area may wary with the distance x from the wall.

# CHAPTER 2

# 2. LITERATURE SURVEY

Myhren et al. [3] investigated the introduction of more heat transferring surfaces to improve the thermal efficiency of the ventilation radiator. The investigation was made using Computational Fluid Dynamics (CFD) simulations while analytical calculations were used for verification of different flow and heat transfer mechanisms. Results showed that heat transfer can be increased in the section where ventilation air is brought into the room by slightly changing the geometry of the fins, decreasing the fin to fin distance and cutting off a middle section of the fin array. This change in internal design could mean considerable increase in thermal efficiency for the ventilation radiator as a whole.

Naphon et al.[4] studied the thermal performance and pressure drop of the helical-coil heat exchanger with and without helical crimped fins are studied. The heat exchanger consisted of a shell and helically coiled tube unit with two different coil diameters. Each coil was fabricated by bending a 9.50 mm diameter straight copper tube into a helical-coil tube of thirteen turns. Cold and hot water are used as working fluids in shell side and tube side, respectively. The experiments were done at the cold and hot water mass flow rates ranging between 0.10 and 0.22 kg/s, and between 0.02 and 0.12 kg/s, respectively. The inlet temperatures of cold and hot water were between 15 and 25 °C, and between 35 and 45 °C, respectively. The cold water was entering the heat exchanger at the outer channel flows across the helical tube and flows out at the inner channel. The hot water entered the heat exchanger at the inner helical-coil tube and flows along the helical tube. The effects of the inlet conditions of both working fluids flowing through the test section on the heat transfer characteristics were discussed.

Bahadori et al.[2]attempted to formulate a novel and simple-to-use correlation for the prediction of efficiencies for uniform thickness finned tubular sections as well as fin tip temperature for wide range of conditions(covering finned pipe diameter to pipe diameter ratios of up to 3).Secondly, another simple correlation was developed to approximate external convection heat transfer coefficients for nominal pipe size(NPS)steel pipes of 75,100,and150 mm arrange in staggered rows surrounded by combustion gases for temperature up to $600^0$C and gas mass flow rates of up to 3kg/$m^2$ s.

Gori et al.[5] conducted experiments in the cooling of an externally finned cylinder with a submerged slot jet of air. Two slots were employed with D/H equal to 2 and 4, where D is the diameter of the cylinder without fins and H the slot height. Local and mean Nusselt numbers were evaluated at several Reynolds numbers and distances from the slot exit. Empirical expressions were proposed to correlate the experimental mean Nusselt numbers and the convective heat transfer coefficients. The two slots were compared also according to the concept of efficiency, which takes into account the cooling rate and the mechanical power necessary to drive the flow.

# CHAPTER 3

# THEORY

## 3.1 INTRODUCTION TO CFD [6]

Computational fluid dynamics, abbreviated as CFD, is a part of fluid mechanics that uses mainly numerical methods and computerized algorithms to solve and analyze problems that involve the flow of fluids. Computers are being used to do the calculations required to simulate the interaction of fluids with surfaces that are defined by boundary conditions, and initial conditions.

The Navier stokes equations form the basis of all CFD problems. In case of CFD, the geometry of the problem is first made. Then the volume of the fluid is quantified into discrete and definite cells which may be referred as the mesh. Then the modeling equations are all set up, boundary conditions defined. The simulation is then done iteratively so that the solution converges to a point. CFD may be used for both steady state and transient state analysis.

Some of the discretization methods used in CFD are as follows:-

1. Finite volume method
2. Finite element method
3. Finite difference method
4. Boundary element method

## 3.2 HEAT TRASNFER BY NATURAL CONVECTION [1]

Heat transfer by natural convection occurs in many places. Let us take a small example when a hot body is placed in contact of a fluid, the density of the fluid varies because of the temperature of the body. The density decreases with increase in temp of the fluid. Free convection or natural convection occurs under the force of gravity. Consider the air that is in the contact with the hot body the fluid in contact gets heated becomes lighter an rises while the cold air takes its position. Again the air gets heated and the cold air takes its places this causes convection loop to be step in the air surrounding the body. Air particles gain velocity owing to the buoyancy force.

3.3 DIMENSIONLESS PARAMETERS OF NATURAL CONVECTION [1]

The continuity, momentum and energy equation are given below:-

Continuity: $\frac{\partial u}{\partial x} + \frac{\partial v}{\partial x} = 0$

Momentum: $\rho(u\frac{\partial u}{\partial x} + v\frac{\partial u}{\partial x}) = -\rho g - \frac{\partial p}{\partial x} + \mu\frac{\partial 2y}{\partial x2}$

Energy: $\rho\, c_p(u\frac{\partial T}{\partial x} + v\frac{\partial T}{\partial y}) = k\frac{\partial 2T}{\partial y2}$

Dimensionless numbers governing the type of flow:

1. Grashoff number $= \frac{g\beta L^3 (T_w - T_\infty)}{V^2}$

   Grashoff number represents the ratio of buoyancy force to the viscous force acting on the fluid.

2. Rayleigh number$= \frac{g\beta L^3 (T_w - T_\infty)}{V\alpha}$

   Rayleigh number is the product of Prandtl number and Grashoff number.

3.4 THE BOUSSINESQ MODEL

For natural convection flow the Boussinesq model is used. In this model fluid density varies as a function of temperature. This model assumes density as a constant value in all solved equations, except for the buoyancy term in the momentum equation:

$$(\rho - \rho_0) = (-\rho_0 \beta(T - T_0)g$$

Limitation of this model is that it cannot be used in cases where the temperature difference is very large.

3.5 FIN PARAMETERS [1]

3.5.1 FIN PERFORMANCE- it may so happen that even after adding fins to a heated surface the heat transfer from that surface does not increase. This may happen because the fins themselves offer a resistance conductive in nature to the heat flow. It can be assesed by

calculating the fin effectiveness which is the ratio of heat transfer with fin to the heat transfer without fin. Usually the use of fins is encouraged only if the value of effectiveness is $\geq$ 2.
It is denoted by the symbol $\varepsilon_f$

$$\varepsilon_f = \frac{heat\ trasfer\ rate\ with\ fin}{heat\ transfer\ rate\ without\ fin}$$

3.5.2 FIN EFFICIENCY – the thermal performance of a fin may be estimates by this parameter. It is denoted by the symbol $\eta_f$. it is the ratio of actual heat transfer from the fin to the maximum heat transfer from the fin if the entire fin was maintained at the fin base temperature.

$$\eta_f = \frac{actual\ heat\ trasfer\ from\ the\ fin}{maximum\ heat\ transfer\ from\ the\ fin\ if\ the\ entire\ fin\ suraface\ is\ ate\ fin\ base\ temperature}$$

# CHAPTER 4

# 4.1 RESULTS AND DISCUSSIONS

## 4.1.1 OPERATING CONDITIONS

The tube wall was maintained at a temperature of 380 K. The enclosure wall were maintained at 300 K ambient temperature. The tube upper surface and lower surface were also maintained at 380 K.

## 4.1.2 MATERIAL PROPERTIES

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Material: air (fluid)

   Property                       Units      Method       Value(s)
   ---------------------------------------------------------------
   Density                        kg/m3      boussinesq   1.225
   Cp (Specific Heat)             j/kg-k     constant     1006.43
   Thermal Conductivity           w/m-k      constant     0.0242
   Viscosity                      kg/m-s     constant     1.789401e-05
   Molecular Weight               kg/kgmol   constant     28.966
   Thermal Expansion Coefficient  1/k        constant     0.003334
   Speed of Sound                 m/s        none         --

Material: aluminum (solid)

   Property               Units    Method     Value(s)
   ---------------------------------------------------
   Density                kg/m3    constant   2719
   Cp (Specific Heat)     j/kg-k   constant   871
   Thermal Conductivity   w/m-k    constant   202.4
```

### 4.1.3 GEOMETRY OF FINS AND TUBE

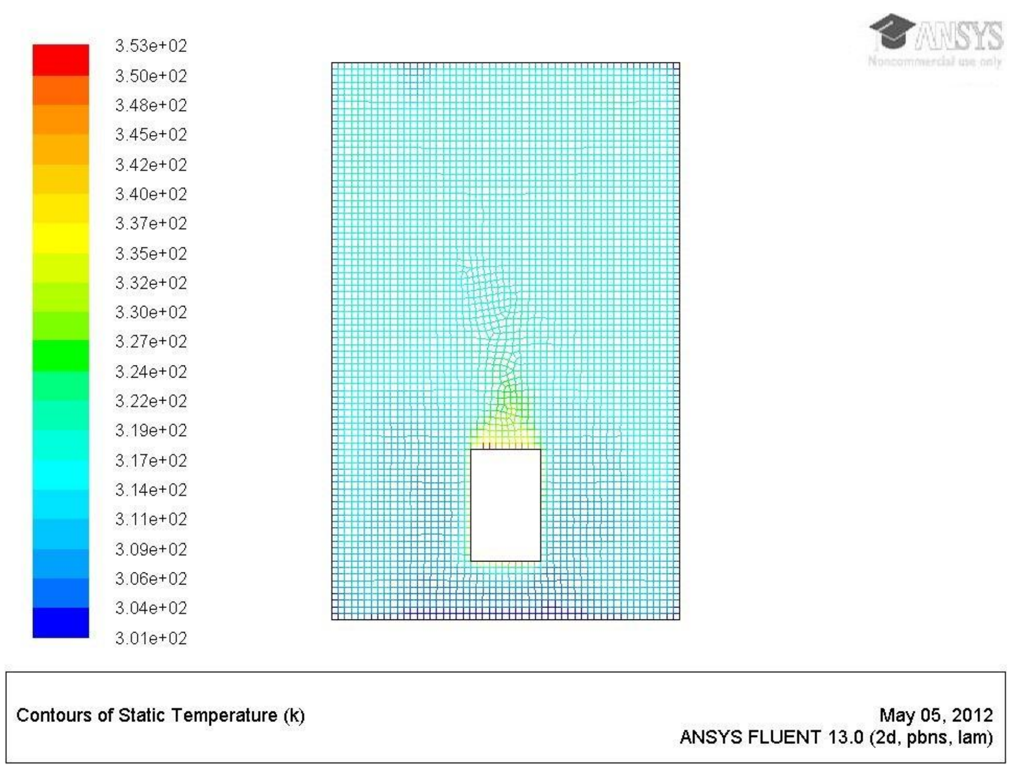
| SL no | Diameter of the tube | Fin height | Fin length | Pitch |
|---|---|---|---|---|
| 1. Tube with no fins | 50 mm | ------ | ----- | ----- |
| 2. Tube with trapezoidal fins | 50 mm | 4 mm (at base)<br>2 mm (at tip) | 5 mm | 4 mm |
| 3. Tube with st. fins | 50 mm | 2 mm | 2 mm | 4 mm |

### 4.1.4 GRAPHS OBTAINED

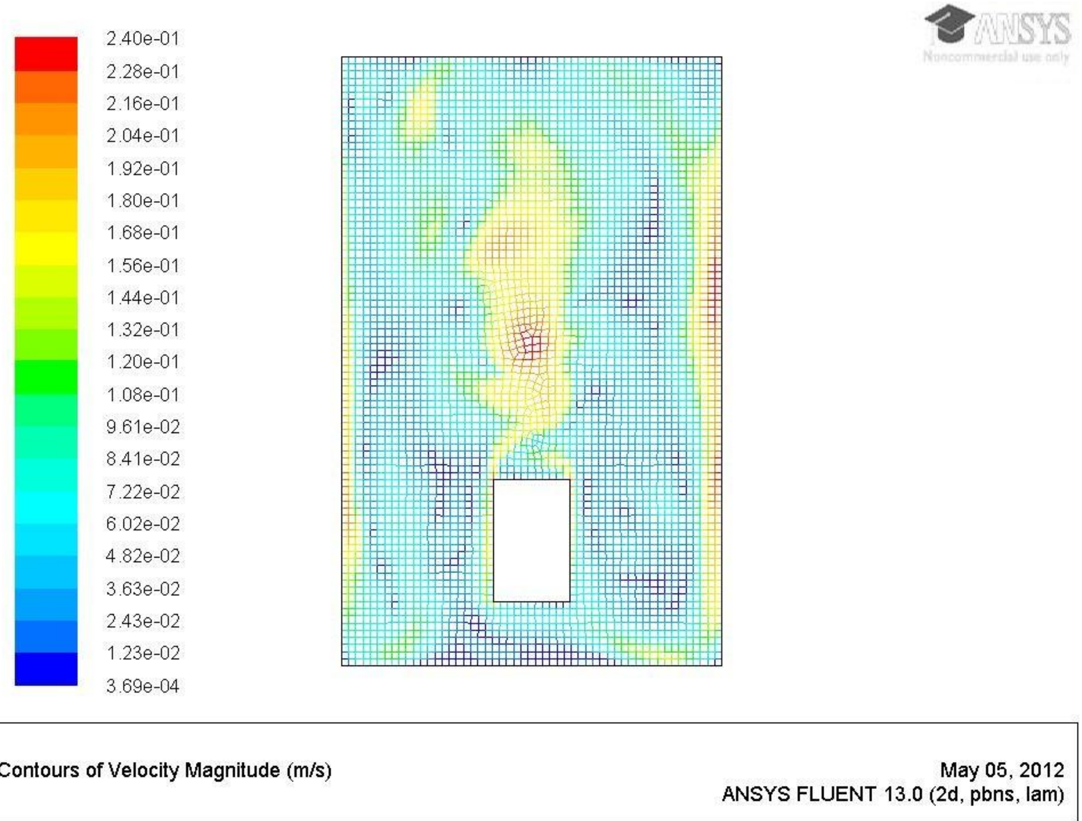
Following are the graphs that were obtained by simulation on ansys 13.0 for various fin configurations

1. Case 1: the tube wall has no fins

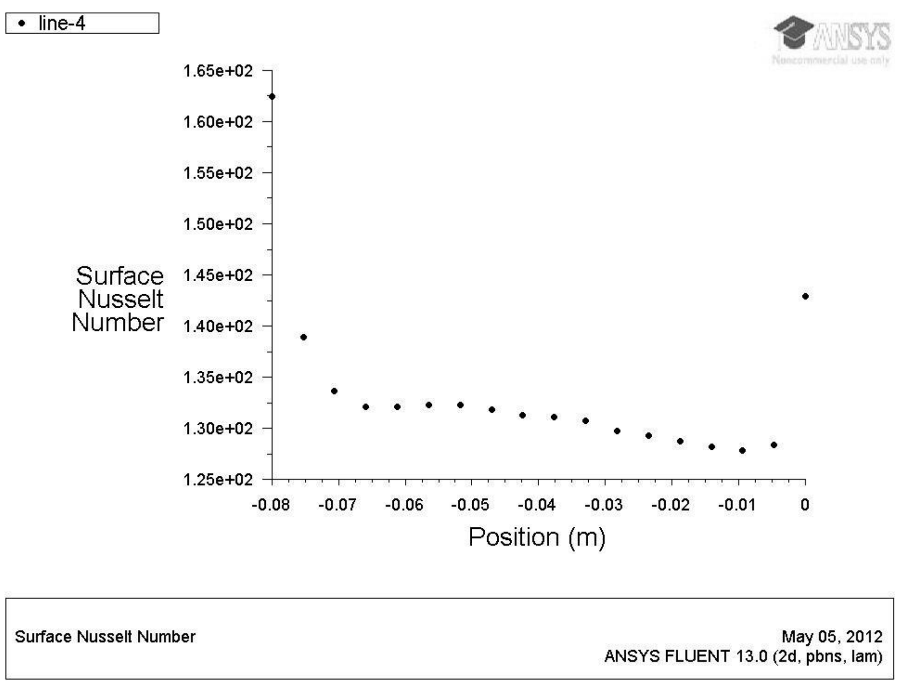
(A) Temperature contours



(B) Velocity vectors

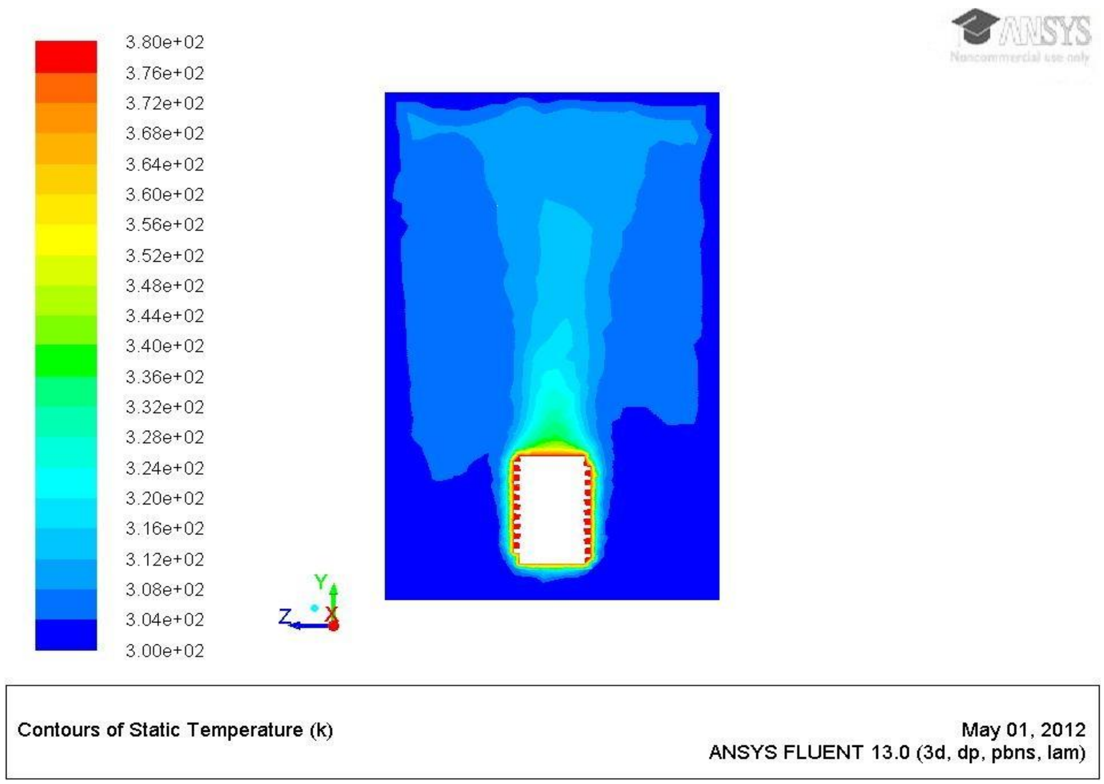



(C) Nusselt number plot on the tube surface

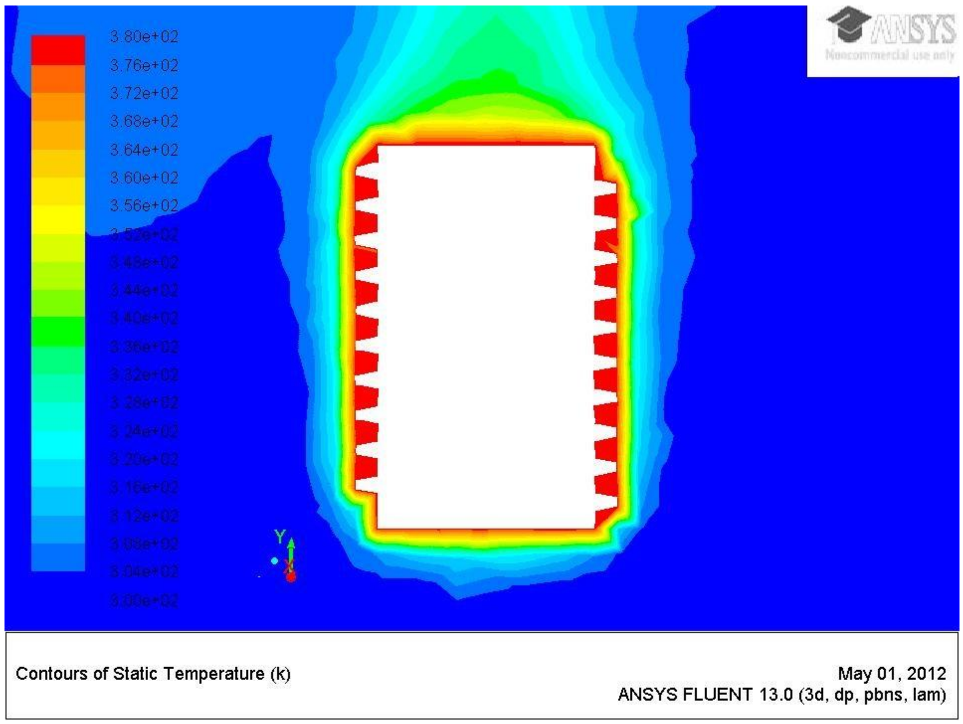


2. Case 2: the tube wall has trapezoidal fins

(A) Temperature contours

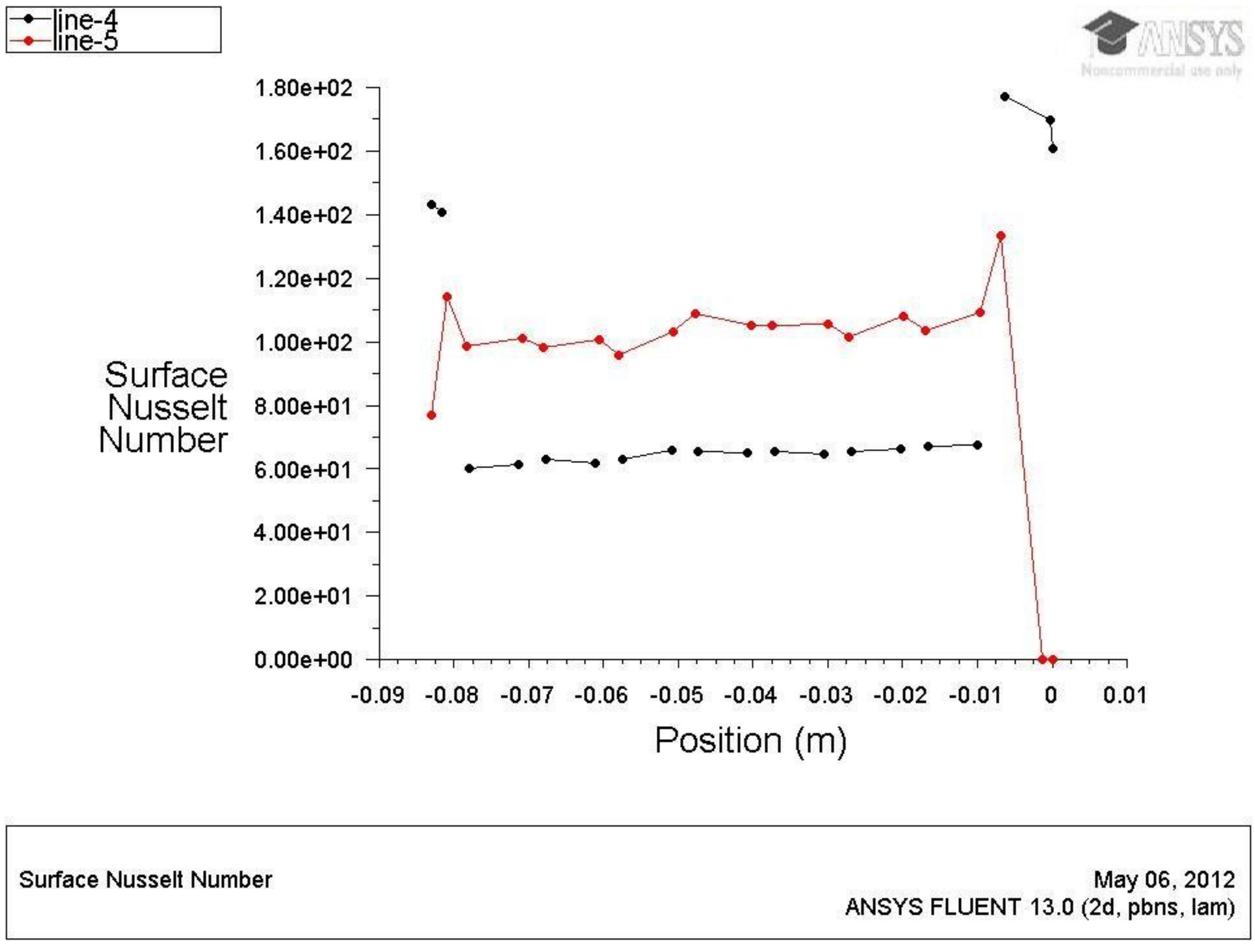



Here is the zoomed view of the temperature contours around the fins



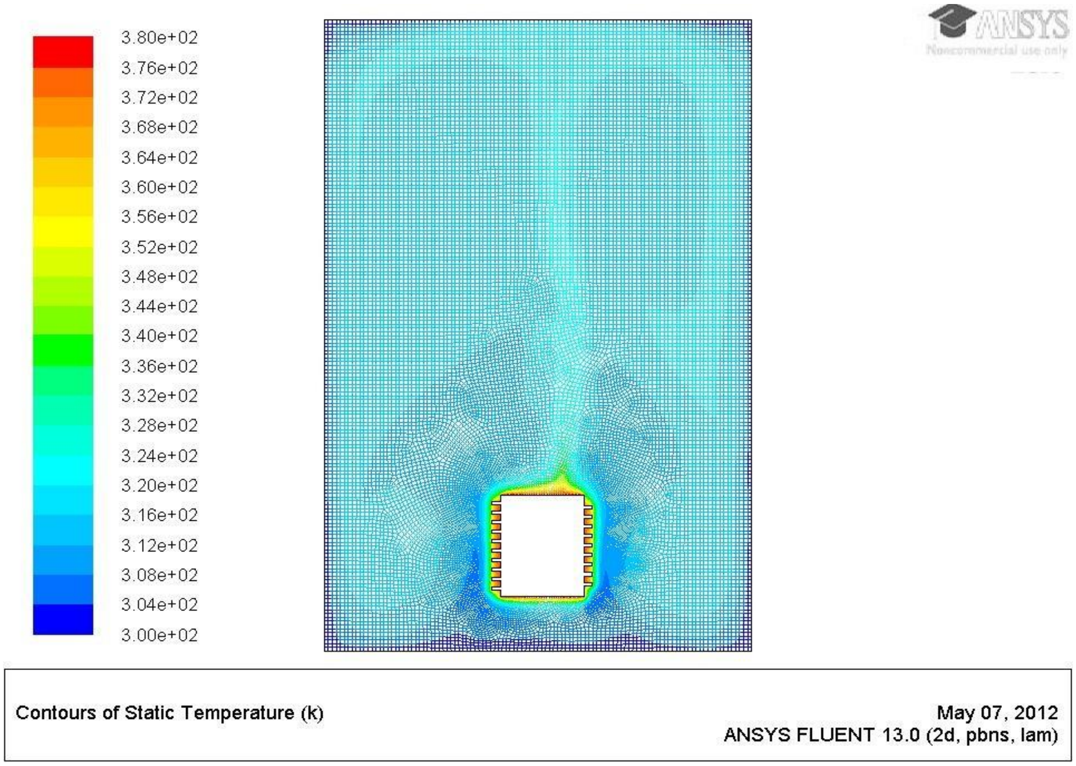
(B) Nusselt number plot on the fin surface and the tube surface

Line 4 is along the fin tip and line 5 is along the tube wall

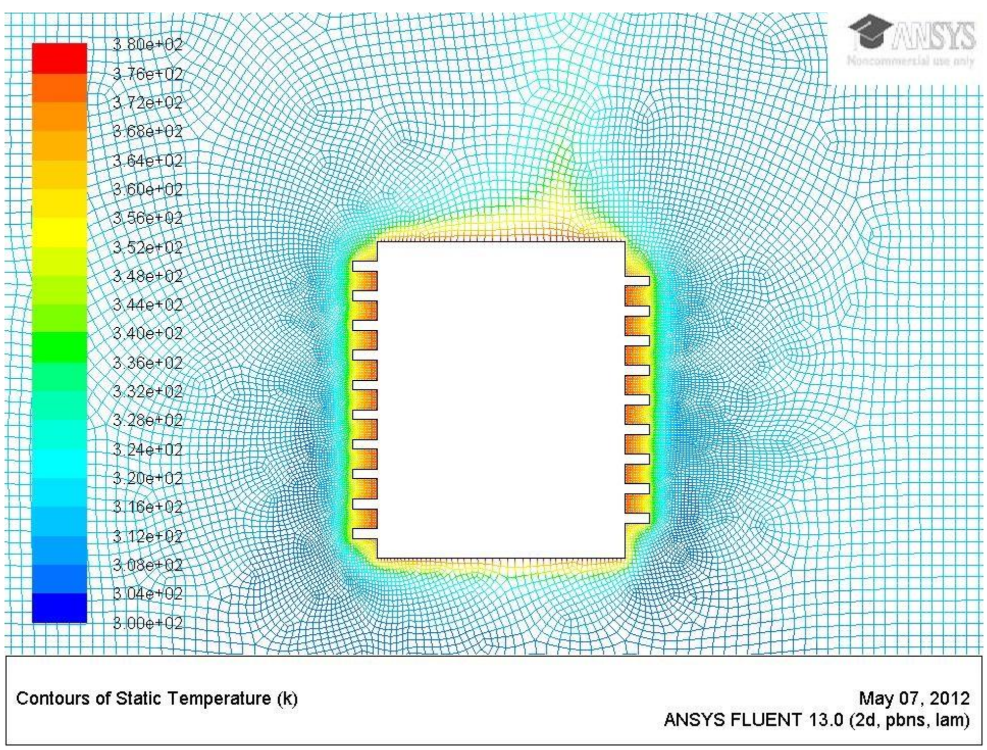


3. Case 3: Tube with straight rectangular fins

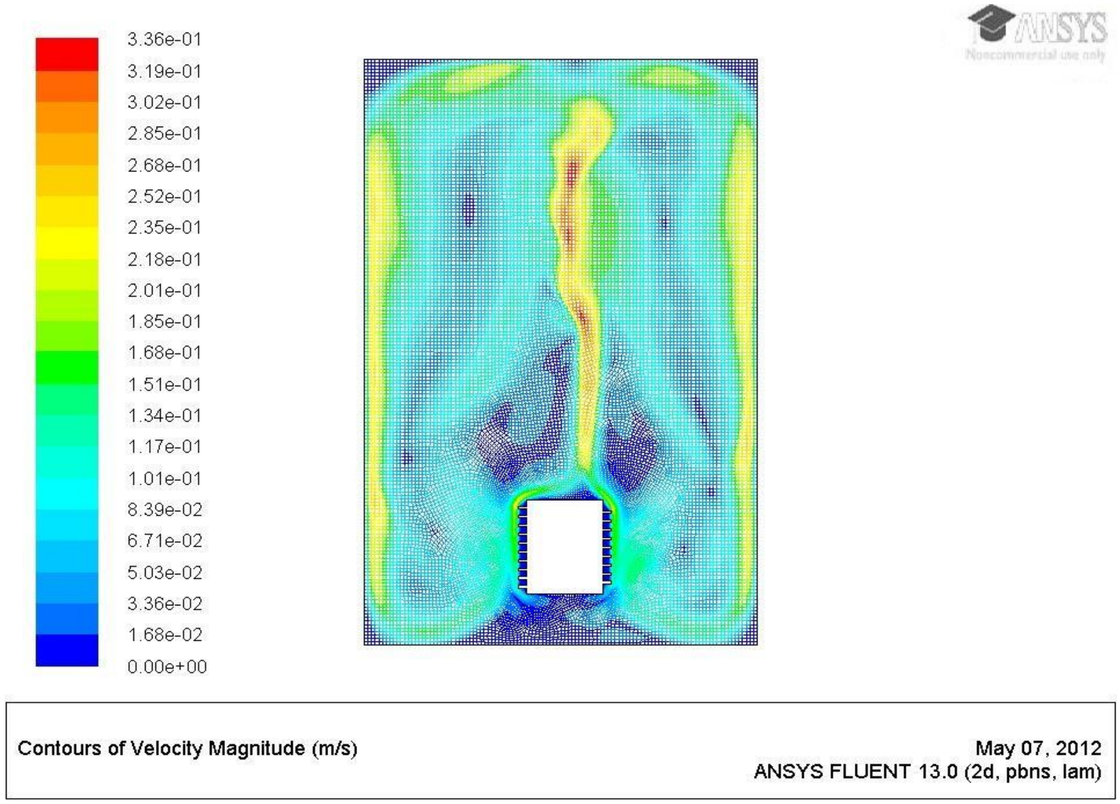
(A) Temperature contours




Here is the zoomed in view of the contours

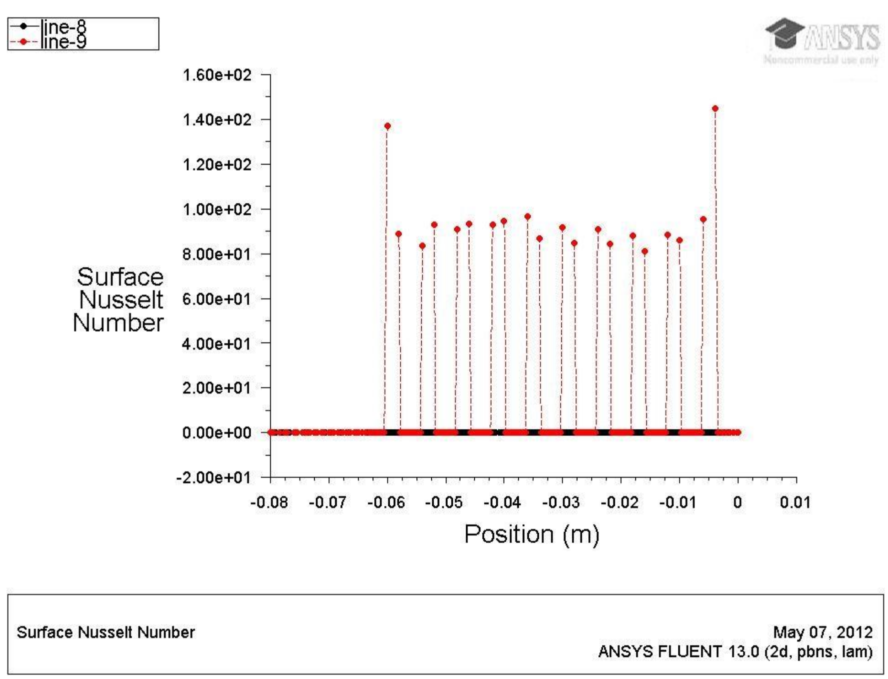


(B) Velocity vectors




(C) Nusselt number plot along the fin tip and the tube wall

Line 8 is along the tube wall and line 9 is along the fin tip



## 4.1.5 TOTAL HEAT TRANSFER

| SL NO | | HEAT TRASFER RATE (W) | |
|---|---|---|---|
| 1. | No fins | wall1(side walls) | 43.995014 |
| | | wall2(top and bottom) | 38.19244 |
| | | Net | 82.187454 |
| 2. | Trapezoidal fins | fins | 45.6348 |
| | | wall | 46.270599 |
| | | Net | 91.905399 |
| 3. | Straight fins | fins | 49.984829 |
| | | wall | 34.575317 |
| | | Net | 84.560146 |

# CHAPTER 5

# CONCLUSIONS

From the graphs obtained above and the total heat transfer rate calculation we can clearly conclude that the fin geometry with trapezoidal profile fins are the most effective in increasing the heat transfer from the tube. The nusselt number values in this case were around 140-145, which is the highest

The second case with the rectangular fins also increased the heat transfer rate but not to a large extent. The nusselt number values in this case were around 130-140.

While in the case of tube with no fins the nusselt number values could reach upto only 120.

Hence from the above anaylsis we can conclude that the to effectively increase the heat transfer rate in heat exchangers or air cooled condenser the profile of fins to be chosen should be trapezoidal.

## NOMENCLATURE

Q- Heat liberated

h- Convective heat transfer coeff

A- Heat transfer area

$T_w$- temperature of the surface

$T_{amb}$- ambient air temperature

$T_\infty$- free stream temperature

T- temperature

ρ- density

g- acceleration due to gravity

u,v – velocity

L- characteristic length

V- velocity

μ- viscosity

β- constant

$c_p$- specific heat capacity

α- constant

# REFERENCES

[1] Nag P.K. ,Heat and mass trasnfer,New delhi:Tata mcgraw hill education private limited, 2010

[2] Bahadori Alireza , Vuthaluru Hari .B ,Predictive tool for estimation of convection heat transfer coefficients and efficiencies for finned tubular sections, International Journal of Thermal Sciences 49(2010) 1477-1483

[3] Myhren Jonn Are, Holmberg Sture ,Improving the thermal performance of ventilation radiators -The role of internal convection fins, International Journal of Thermal Sciences 50 (2011) 115-123

[4] Naphon Paisarn ,Thermal performance and pressure drop of the helical-coil heat exchangers with and without helically crimped fins, International Communications in Heat and Mass Transfer 34 (2007) 321–330

[5] Gori F., Mascia M., Petracci I. Air cooling of a finned cylinder with slot jets of different height,International Journal of Thermal Sciences 50 (2011) 1583-1593

[6] http://en.wikipedia.org/wiki/Computational_fluid_dynamics

[7] http://jullio.pe.kr/fluent6.1/help/html/ug/node1283.html